Modal Analysis of Gear Housing and Mounts

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ABSTRACT

Dynamic finite element analysis of a real gear housing is presented. The analysis was conducted for the housing without the rotating components (gears, shafts, and bearings). Both rigid and flexible mounting conditions for the gear housing are considered in this analysis. The flexible support simulates the realistic mounting condition on a rotorcraft, and the rigid one is analyzed for comparison purposes. The effect of gear housing stiffeners is also evaluated. The results indicate that the first six natural modes of the flexibly mounted gear housing in the 0 to 200 Hz range correspond to the translational and rotational rigid body vibration modes of the housing. Above this range, the housing plate elastic modes begin to occur. In the case of the rigid mount, only the housing plate elastic modes are observed which are verified by modal analysis experiments. Parametric studies show that the housing plate stiffeners and rigid mounts tend to increase most of the natural frequencies, the lower ones being affected the most.

INTRODUCTION

Gear noise and vibration is a major problem in many power transmission applications as is evident from the literature (refs. 1 to 8). This problem becomes more significant in applications with higher operating speeds where the vibratory excitation, which is related to the gear transmission error (refs. 7, and 9 to 11), occurs at frequencies in the order of several kilohertz. Most of this vibratory energy generated at the gears is transmitted to the housing and attached structures through the structure-borne paths involving the shafts, bearings, and mounts (refs. 4, 5, 7 and 12). In rotorcraft applications the vibratory energy transmitted from the gears to the aircraft structure results in a high level of cabin noise. Hence, a fundamental knowledge of the dynamic behavior of a gearbox and its supporting structure is essential in the overall goal of reducing gear initiated vibration problems.

In this study, the dynamic properties of a real gear housing are predicted using the finite element method (FEM) and verified using experimental modal analysis (EMA). The finite element program ANSYS (ref. 13) was used to calculate the natural frequencies and modes of a real gear housing and mounts without the gears, shafts and bearings. A description of the actual gearbox and the FEM model used to model it are given. Results of the FEM analysis as compared to the experimental modal analysis results for the rigidly mounted

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stiffened gearbox case are also presented. Finally, results are given on the effects of the mounts and stiffeners on the dynamics of the housing, as predicted by the FEM model.

FINITE ELEMENT MODEL

The FEM model was constructed to simulate the experimental gearbox at NASA Lewis Research Center. The simulated gearbox, as shown in figure 1, is approximately 0.254 by 0.279 by 0.330 m (10.0 by 11.0 by 13.0 in.), and all of its plates are 0.006 m (0.25 in.) thick made of 1020 steel. The regions near the bearings are 0.025 m (1.0 in.) thick. There are four circular holes for the bearings, two at each side plate supporting the shafts. Figure 2 illustrates the 0.254 m (10.0 in.) tall flexible mount frame which is constructed from eight 0.006 m (0.25 in.) thick, 1020 steel angle beams. The fuselage sheet is a 0.006 m (0.25 in.) thick aluminum plate of dimensions 0.762 by 0.660 m (30.0 by 26.0 in.), and is attached horizontally to the flexible mount structure. The housing is supported at each corner of the base plate for all mounting conditions, and the mounts are attached to a rigid foundation.

The FEM models of the rectangular gearbox with the gears, shafts, and bearings removed are shown in figure 3 for the rigidly and flexibly mounted housings. These FEM models consist of four-noded quadrilateral plate elements with bending and membrane capabilities for the housing and fuselage attached, and two-noded beam elements with shear deformation and rotary inertia capabilities for the flexible mount skeleton and housing plate stiffeners. The boundary conditions are: (1) zero displacements and rotations at each corner of the base plate for the rigid mount and (2) similar conditions at each foot of the flexible mount. The interface between adjacent housing plates are assumed to be continuous. About 100 dynamic degrees of freedom are specified to minimize computational effort while still maintaining sufficient accuracy. Natural frequencies are extracted up to at least 4 kHz to include the gear mesh frequency regime.

EXPERIMENTS AND MODEL VALIDATION

Modal experiments were performed on the NASA high precision gearbox with the gears, shafts, and ball bearings installed. An approximate configuration of the NASA gearbox is shown in figure 4. The nominal dimensions of the gear housing have been given in the previous section. The variable center distance gear-shaft pair is supported by four ball bearings. The four side plates and base plate are welded together, while the top plate is bolted to the side plates. The housing plates are stiffened internally, and the gear housing system is mounted rigidly to a massive foundation. Dynamic transfer functions were obtained only on the exterior of the gear housing structure using a two channel dynamic signal analyzer. For these experiments, 154 degrees of freedom were selected in the direction transverse to the plane of the housing plate, with the reference point being approximately near the center of the top plate to avoid nodal points of interest. Natural frequencies and modes were estimated using a commercially available modal analysis program. Here, the exponential method was used to extract modal parameters and generate analytical functions for the transfer functions, while the circle fit method was used to construct the modal vectors.

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The FEM model predictions were compared with EMA tests performed on the NASA experimental gearbox. Table I lists the resulting natural frequencies and mode shapes. As depicted in this table, FEM predictions are in good agreement with EMA. The results in table I also suggest that the gears, shafts, and bearings present in the actual housing have no appreciable affect on the dynamics of the housing, since the FEM modeled the housing without these components. This may not be true for the flexibly mounted case, where the added mass and stiffness of the gears, shafts, and bearings could result in an appreciable difference in the rigid body modes of the actual system as compared to the FEM predictions. Mode shapes predicted by FEM, for the rigid mount case, are illustrated in figures 5 and 6. Identification of a mode shape is based on its most dominant feature due to its complexity. For each mode, two simplified illustrations are shown: (1) mode shape of the three visible plates, and (2) mode shape of the three nonvisible plates in an approximate isometric view. Figure 5 compares the simplified mode shape to the actual one for the third mode. The first mode is basically a first transverse mode of the top housing plate with small amplitudes of vibration at the side plates. Second to fourth modes are given by different combinations of first transverse housing plate modes. For example, in the second mode, the side plates supporting the bearings are in-phase (i.e., both plates are moving in the same direction with respect to the global coordinates). Third and fourth modes are given by out-of-phase transverse vibration of these side plates, but with different combinations for the other plates. The fifth mode constitutes a second transverse plate mode at the top plate combined with first and second transverse plate modes at the side plates. The higher modes, not shown here, are also given by similar combinations of plate modes. Comparison of the higher modes are made on the basis of modal density for one-third octave band center frequencies in the 500 to 4000 Hz range, as shown in figure 7, because of the high number of participating modes observed. The results again indicate that FEM is in good agreement with EMA.

PARAMETRIC STUDIES

Effects Of Mounts

The FEM model was used to investigate the effects of mounting flexibility on the dynamics of the gear housing system. The rigidly mounted gear housing is observed to possess only housing plate elastic modes. On the other hand. FEM model of the flexibly mounted gear housing indicates that the first six natural modes are the translational and rotational rigid body modes of the gear housing as shown in figure 8. For example, the first natural frequency at 54 Hz corresponds to the gear housing vibration in the Y-direction as shown. In addition, the natural frequencies are considerably lower, by approximately one order of magnitide, as compared to those of the rigidly mounted gear housing. These rigid body vibration modes result from the complex elastic deformation of the flexible mount skeleton and fuselage sheet. The housing plate natural frequencies are also lowered, especially the first few, when the box is mounted flexibly. Figure 9 compares the modal density for one-third octave band center frequencies of the flexibly mounted gear housing to that of the rigidly mounted one. The modal density at frequencies above 1 kHz is seen to be higher for both conditions. Figure 10 compares the natural frequencies of the flexible mount, rigidly mounted housing, flexibly mounted housing, and the gear-shaft system. The modal distribution of the gear shaft system is seen to be significantly lower than that of the housing and mounts. The

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natural frequencies of the gear shaft system were added in figure 10 to compare the modal densities of the major components of the test gearbox. Here it is assumed that the modes of the gear shaft system are independent of the housing plate modes; coupling issues are currently being investigated. As expected, the natural frequencies of the flexible mount are lowered when the housing is added to it, as shown in figure 10.

Effect Of Stiffeners

The introduction of housing plate stiffeners, as shown in figure 4, do not change the nature of the mode shapes predictions. However, the natural frequencies for this case are higher as illustrated in figure 11. The lower natural frequencies are affected more by the stiffeners than the higher frequencies. Also, some of the higher modes are suppressed by the stiffeners which reduce the modal density further in the higher frequency bands.

CONCLUSIONS

Based on the analytical and experimental results obtained in this study, the following conclusions can be made:

(1) The FEM model used accurately predicted the dynamic characteristics of an experimental gear housing, as verified with modal analysis experiments on the actual hardware.

(2) The flexibility of the gear housing mount directly influences the type and frequencies of the primary housing modes. In the rigidly mounted case the primary modes correspond to the elastic plate modes of the housing. In the flexibly mounted case the primary modes occurred at frequencies lower than the rigid mounted case, with the mode shapes corresponding to rigid body modes of the housing.

(3) The addition of housing plate stiffeners did not significantly change the nature of the mode shape predictions. However, the mode shapes of the housing with stiffeners occurred at higher frequencies than the corresponding mode shapes of the housing without stiffeners.

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TABLE I. – COMPARISON OF NATURAL FREQUENCY RESULTS

Mode shape[a]	Natural frequency, Hz	
	FEM	EMA
First top plate mode	501	497
In-phase first side plates mode	598	584
Out-of-phase first side plates mode I	627	639
Out-of-phase first side plates mode II	752	781
Second top plate mode	889	877

[a]See fig. 6.
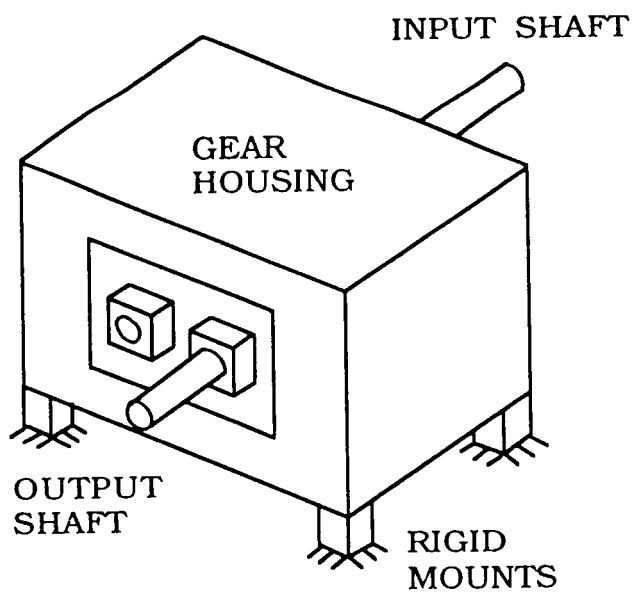
FIGURE 1. - GEARBOX WITH RIGID MOUNTS.

Labels in figure: INPUT SHAFT, GEAR HOUSING, OUTPUT SHAFT, RIGID MOUNTS

FIGURE 2. - GEARBOX ON FLEXIBLE MOUNTS.

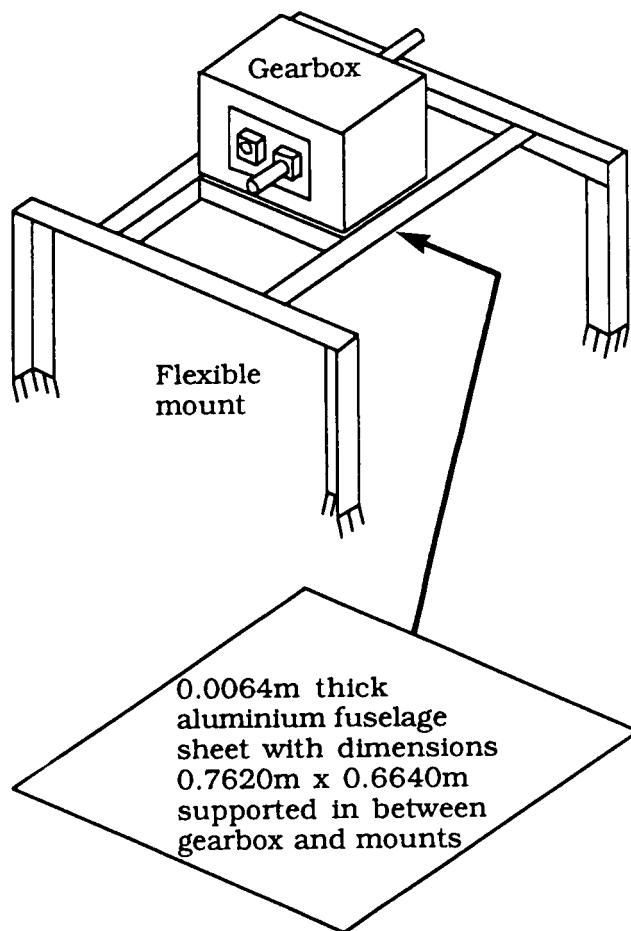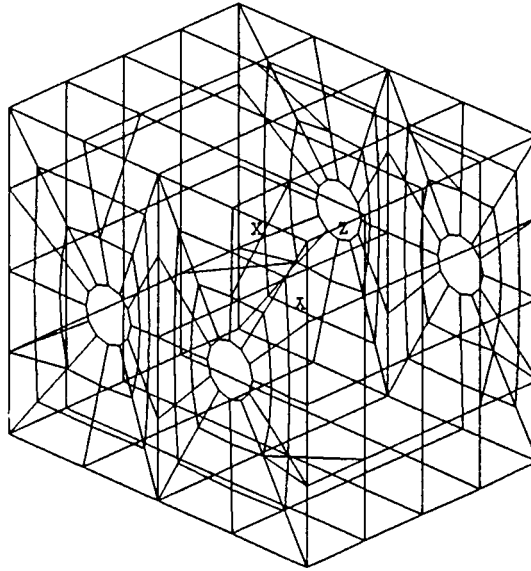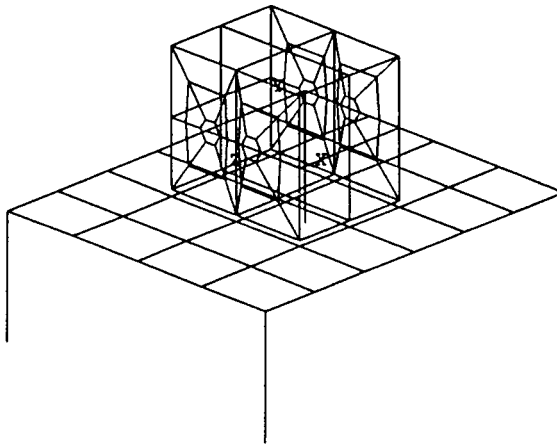Labels in figure: Gearbox, Flexible mount, 0.0064m thick aluminium fuselage sheet with dimensions 0.7620m x 0.6640m supported in between gearbox and mounts

box on rigid mount (Fig. 1)

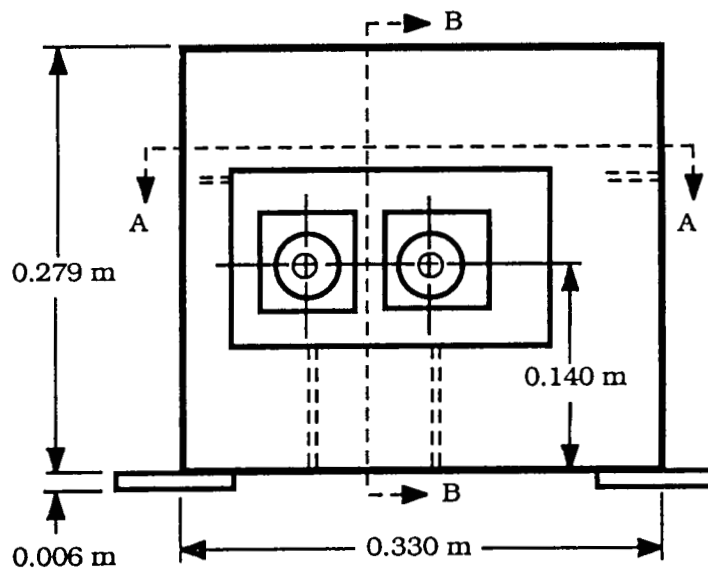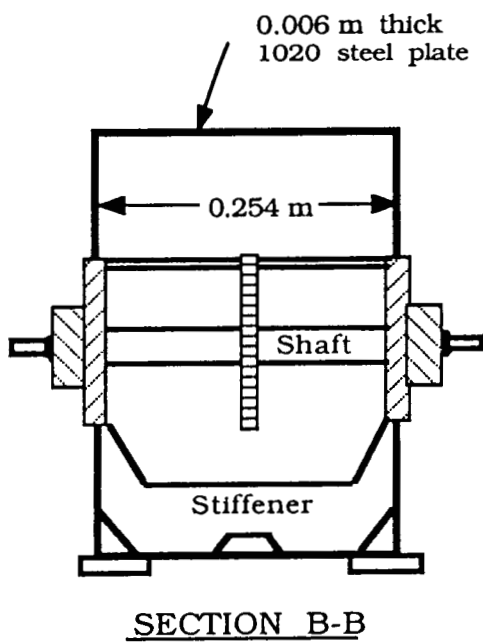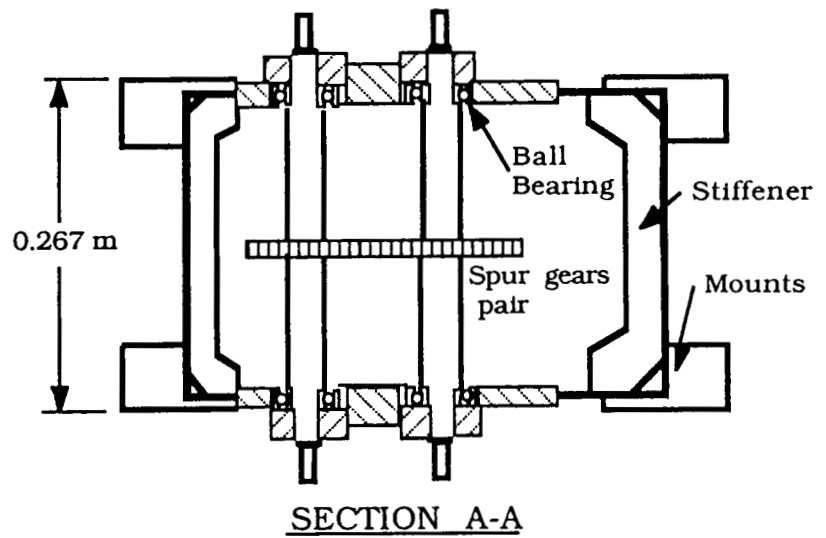box on flexible mount (Fig. 2)

FIGURE 3. - FEM MODELS OF THE GEARBOX.

SECTION A-A

SECTION B-B

FIGURE 4. - SCHEMATIC OF THE NASA GEARBOX.

Actual

visible plates   nonvisible plates
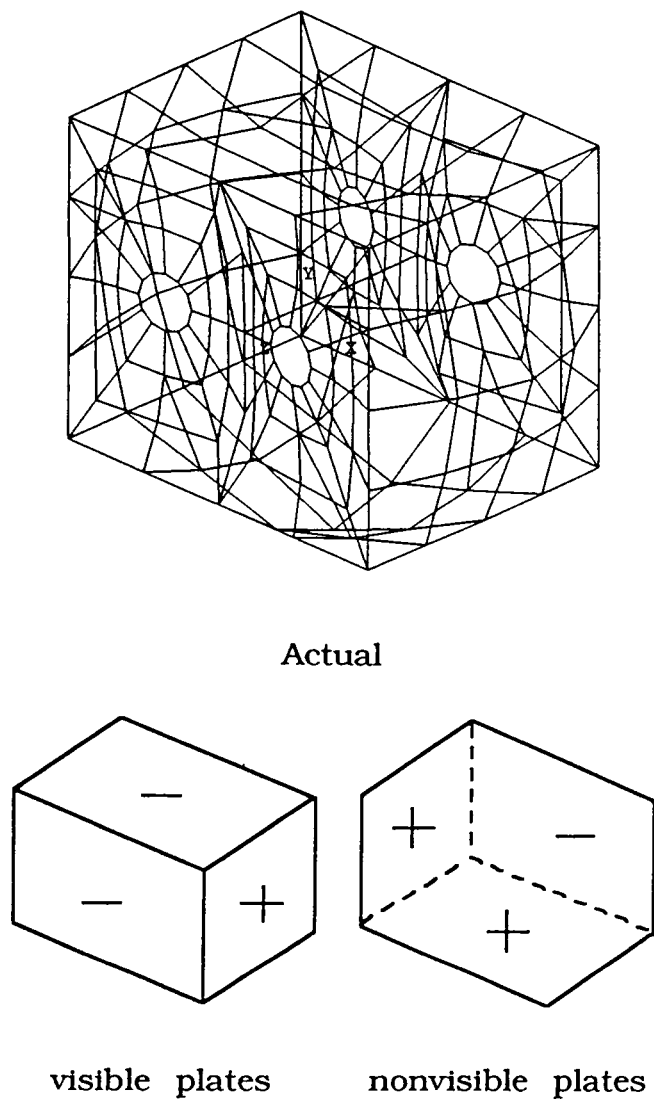
FIGURE 5. - THIRD MODE SHAPE OF THE RIGIDLY MOUNTED GEARBOX.

$f_1 = 501$ Hz $f_2 = 598$ Hz

$f_3 = 627$ Hz $f_4 = 752$ Hz

$f_5 = 889$ Hz
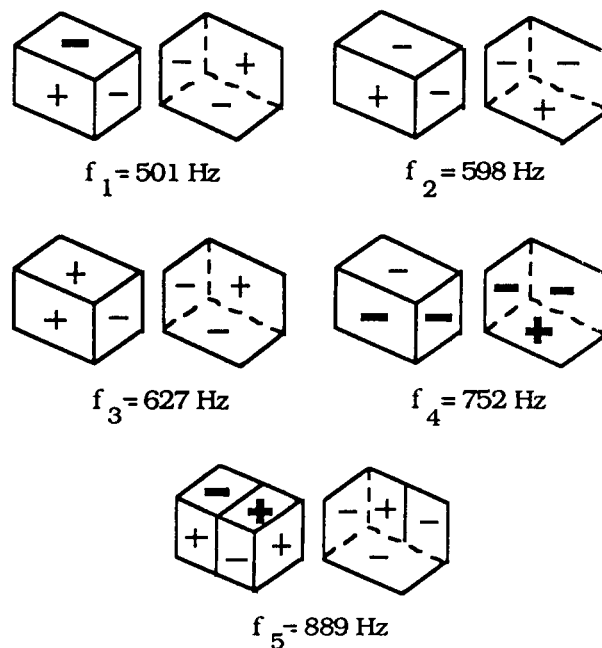
FIGURE 6. - MODE SHAPES OF THE RIGIDLY MOUNTED, STIFFENED NASA GEARBOX SHOWN IN FIGURE 4. BOLD SIGN IMPLIES LARGER AMPLITIDE. ACTUAL BOX ORIENTATION IS THE SAME AS SEEN IN FIGURE 5.
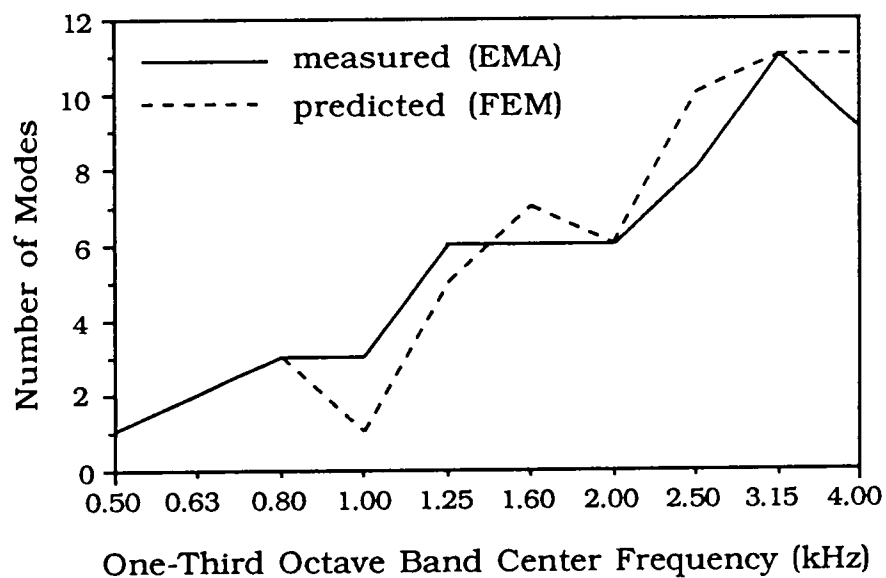
FIGURE 7. - PREDICTED AND MEASURED MODAL DENSITIES.

$f_1 = 54$ Hz

$f_2 = 66$ Hz

$f_3 = 72$ Hz

$f_4 = 93$ Hz
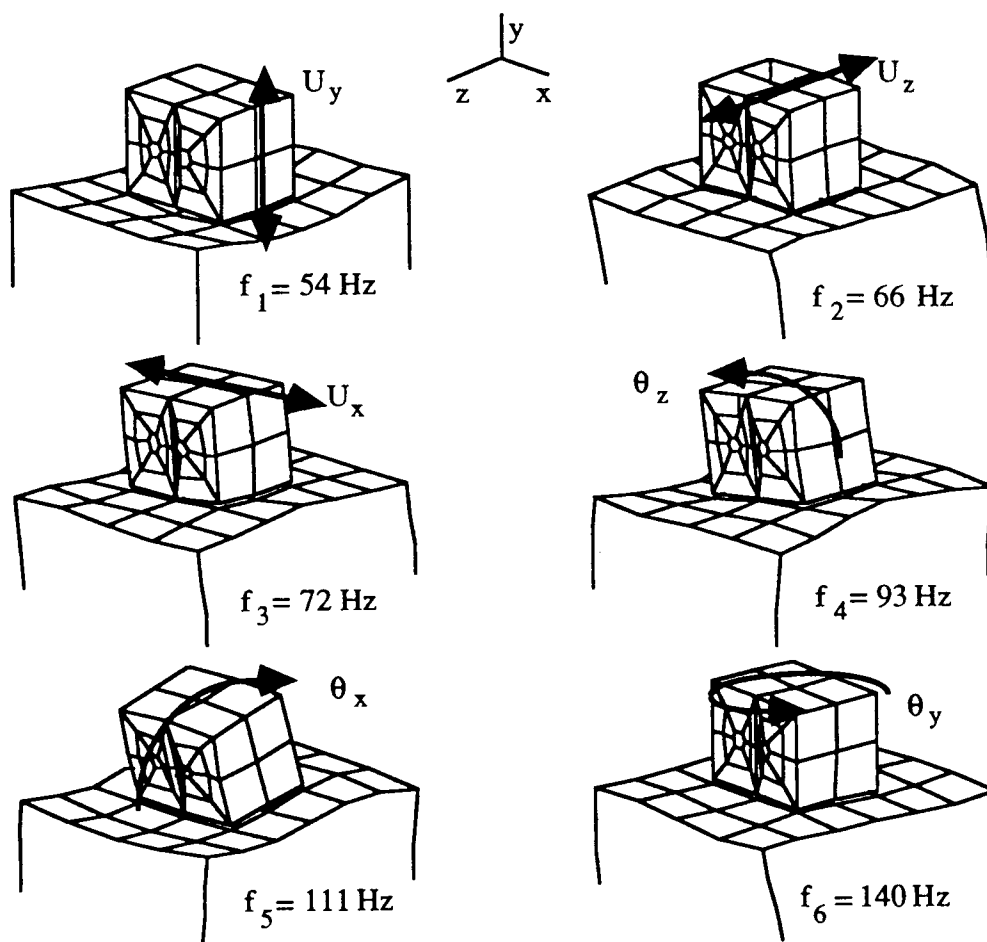
$f_5 = 111$ Hz

$f_6 = 140$ Hz

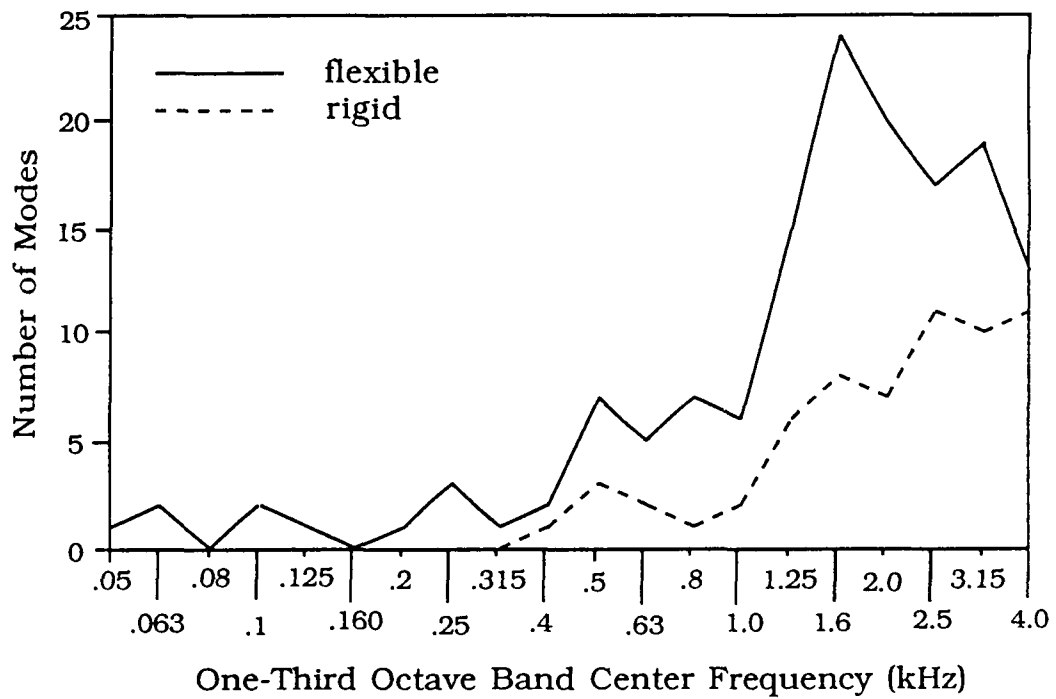FIGURE 8. - RIGID BODY MODES OF THE FLEXIBLY MOUNTED GEARBOX.

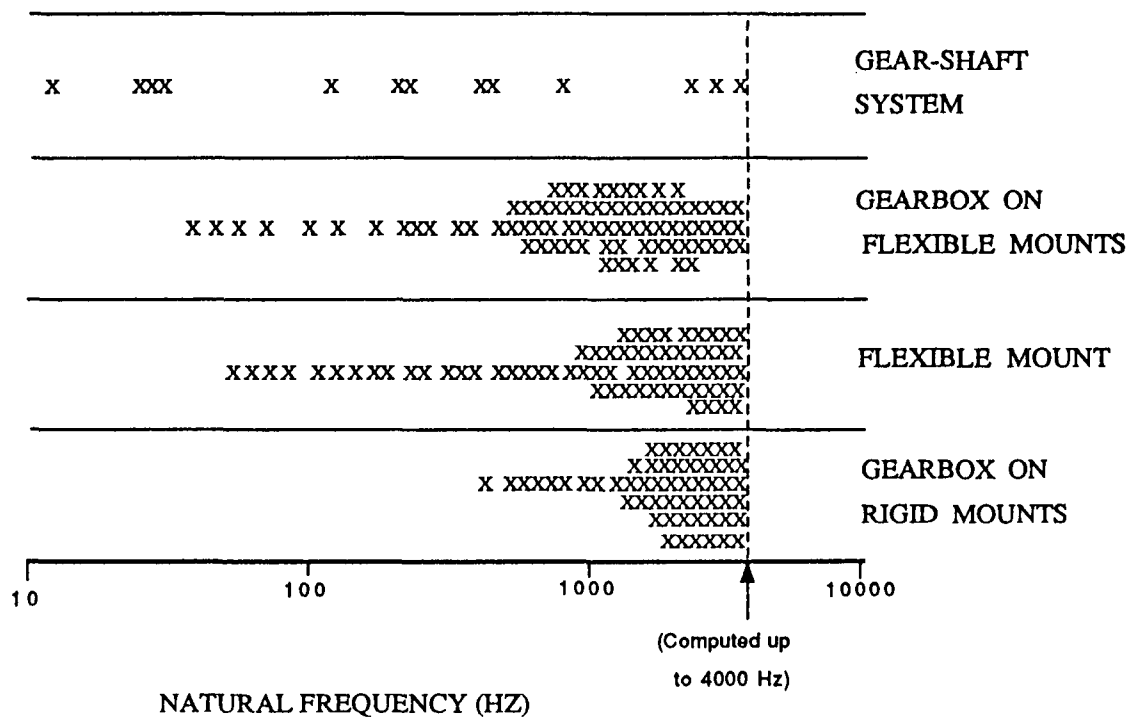FIGURE 9. - MODAL DENSITIES FOR RIGID AND FLEXIBLE MOUNTING CONDITIONS.

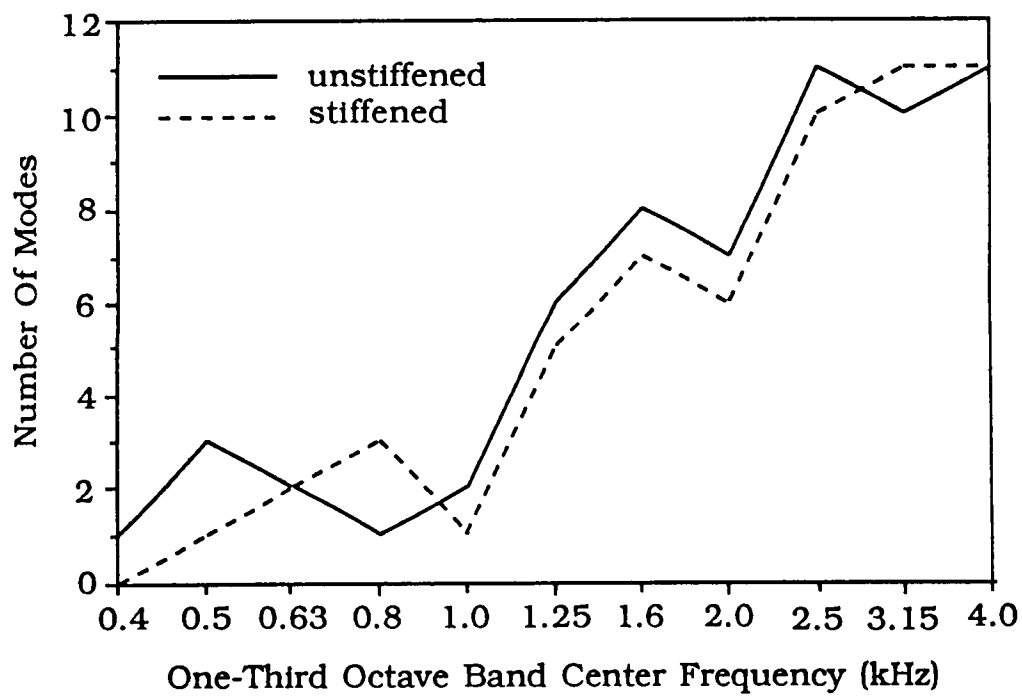FIGURE 10. - MODAL DISTRIBUTION FOR THE MAJOR COMPONENTS OF THE GEARBOX.

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FIGURE 11. - MODAL DENSITIES FOR THE STIFFENED AND UNSTIFFENED, GEARBOX ON RIGID MOUNTS.

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16. Abstract

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